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Characteristics of Mechanical Noise during Motion Control Applications

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1. Introduction

Signal characteristics and processing are an important factor during today's digital world including the motion control and strain measurement applications. A digital signal is someone that can assume only a finite set of values is given for both the dependent and independent variables being analyzed (Smith, 2006). The independent variables are usually time or space; and the dependent variables are usually amplitudes. To use digital signal processing tools effectively, an analog signal must be converted into its digital representation in time space. In practice, this is implemented by using an analog-to-digital converter (A/D), which is an integral part of data acquisition (DAQ) cards (Vaseghi, 2009). One of the most important parameters of an analog input system is the sampling rate at which the DAQ card samples an incoming signal. During the measurement and processing of the signal digitally, it is common to face noise problems interfacing the signals captured (Yuksekkaya, 1999; Chu & George, 1999; Kester, 2004). The noise could be coming from various sources with different characteristics and affecting the measurement systems. Once the signal is contaminated with the noise, the reading from the instruments will not be representing the actual situation of the physical phenomenon being captured. Therefore, it is an important area of practice to analyze the characteristics of the noise for any implementation of the signal analysis before constructing further refinements for data analysis. Furthermore, it would be a more practical to take some precautions in order to reduce the effects of the noise on the signal. Even it is possible to use some tools to decrease the effects of the noise to the signal ratio, it would be more practical to eliminate the noise as much as possible at the first hand. It is also evident from the industrial applications that the cost of initial investments for any noise elimination applications is cheaper than that of later investments.

Computer-based data acquisition systems using small computers have been successfully applied in many industrial applications including the motion control processes producing high performances at relatively low costs. As the investment cost of data processing systems decreases, it is getting more common to see a number of data acquisition systems implemented applications in our daily life. The benefits of a data acquisition system include: an improved analysis, accuracy and consistency, reduced analysis time and cost, and lower response time for an out-of-control situation regarding quality. It could be easily noticed

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that there would be a tremendous amount of noise superimposed on the signal coming from the measurement units. The noise could be coming from different sources depending on the application area. The main sources of the noise, however, are mechanical and electrical noises commonly found at the industrial applications (Yuksekkaya, 1999). Therefore, refinements are necessary for most of the times so that the noise problems could be eliminated from the signal in order to make an accurate measurement during the motion control.

During the industrial applications such as CNC controlled lathes and load cells taking the dynamic measurements, a considerable amount of mechanical noise could be superimposed to the signal from the ground due to the vibration of the buildings. The mechanical noise problem could damage the reading from the instruments due to the noise superimposed to the signal. In this text, an extensive analysis of the mechanical noise due to the building vibration has been analyzed and possible solutions to the problem discussed.

2. Diagnostics of noise in the signal

As stated, digital signal is a finite set of values in both the dependent and independent variables. One of the most important parameters of an analog input system is the rate at which the DAQ card samples an incoming signal. A fast sampling rate acquires more points in a given time. As a result, a better representation of the signal is formed. Sampling too slow may result in a poor representation of the signal. This may cause a misrepresentation of a signal, which is commonly known as an aliasing effect. In order to avoid aliasing effects, the *Nyquist Theorem* states that a signal must be acquired at the rate greater than twice the maximum frequency component in the signal acquired (Ramirez, 1985). Figure 1 indicates the basic divisions of different signal types. The most fundamental division is stationary and non-stationary signals. Stationary signals are characterized by average properties that do not vary with time and independent of the particular sample record used to determine them. The term “non-stationary” covers all signals that do not satisfy the requirements for stationary signals. Computer-based data acquisition systems using small computers have been successfully applied in many applications producing high performances at relatively low costs. The benefits of a data acquisition system include: an improved analysis, accuracy and consistency, reduced analysis time and cost, and lower response time for an out-of-control situation regarding quality. A typical data acquisition system consists of several parts: a signal conditioning module, a data acquisition hardware (A/D converter), analysis hardware, and data analysis software.

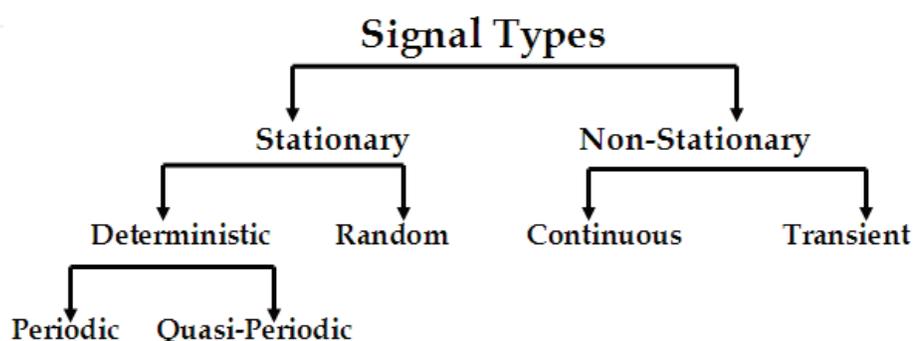


Fig. 1. Classification of signal types.

In any digitally working environment, most of the time, there were two types of main noise problems that needed to be solved in order to have a meaningful result as follows (Yuksekkaya & Oxenham 1999; Yuksekkaya et. al. 2008):

- 1) Electrical Noise:
 - i) Static
 - a) Fluorescent lamps
 - b) Computer screens
 - c) Others
 - ii) Dynamic
 - a) AC power lines
 - b) Stepper motor
 - c) Transformers
 - d) Magnetic fields from other equipment
- 2) Mechanical Noise:
 - i) Vibration from stepper motor
 - ii) Vibration from building
 - iii) Vibration from the other sources

2.1 Electrical noise

Tensile testing devices are a combination of a strain measurement unit and a stepper motor which drives the measurement unit. During the processing of a strain measurement, the location of measurement unit should be precisely located in order to have an accurate stress-strain reading from the instrument. Most of the time, strain measurements are taken in the presence of electrical and magnetic fields, which can superimpose electrical noise on the measurement signals. If the electrical noise is not controlled properly, the noise can lead to inaccurate results and incorrect interpretation of the signals coming from the strain gages as well as the inaccurate location data. In order to control the noise level and maximize the signal-to-noise ratio, it is first necessary to understand the types and characteristics of electrical noise as well as the sources of such noises. Without understanding the noise and its sources, it is impossible to apply the most effective noise-reduction methods on any particular instrumentation problem.

Virtually, every electrical device that generates, consumes, or transmits power is a potential source for causing noise in strain gages. In general, the higher the voltage or current level and the closer the circuit is to the electrical device, the greater the induced noise will be superimposed to the signal. A list of common sources of electrical noise could be found in any signal analysis textbooks and electrical noise from those sources could be categorized into two basic types, that is: electrostatic and magnetic noises (Croft et al., 2006; Agres, 2007). The characteristics of these two types of noise are different; and they require different noise reduction techniques in order to eliminate their effects on the signals. Most of the noise coming from outside may be eliminated by using shielded, twisted cables and eliminating the ground loops in the system (more than one connection of the system to the ground). Furthermore, electromagnetic noises could be eliminated by using a special designed apparatus named as *Faraday Cage* if the application requires it.

2.2 Mechanical noise

It would be practically possible to see that a strain measurement signal could be so sensitive that it would be continuously picking up mechanical noise from different sources such as from the buildings and from the stepper motors. It is necessary to analyze the building and stepper motor vibration sources separately. An extensive analysis of the mechanical noise coming from the building and potential solutions for the vibration sources are given as follows:

2.2.1 Building vibration

Laboratory measurement instruments could be located in either stationary or mobile laboratories depending on the type of measurements necessary to perform. Regardless of the location of the instruments, it is a known fact that the ground vibration will affect the instrument's reading if correct precautions will not be applied. A considerable amount of mechanical noise would be coming from the ground due to the vibration of the building. In order to eliminate, or at least minimize, the effect of the ground movement as much as possible, usually the testing instrument was mounted on the top of a heavy marble block that was supported by a spring-like material. In order to analyze the vibrating mechanical system, let us consider an object hanging from a spring as shown in Figure 2 (Halliday et al., 2007; Zill & Cullen, 2006).

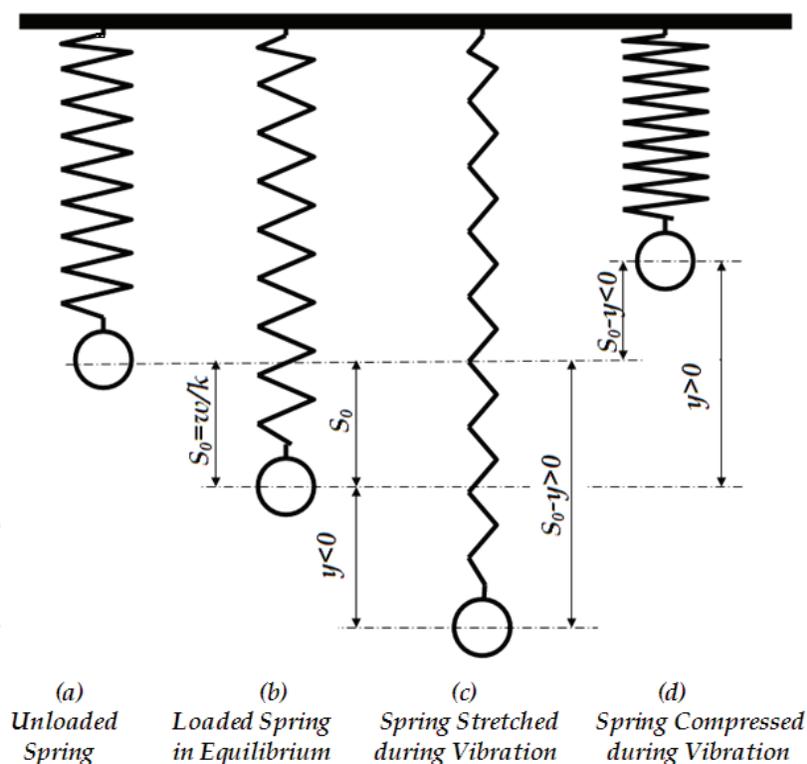


Fig. 2. Schematic of mass spring system.

The weight, w , of the object is the magnitude of the force of gravity acting on the measurement instrument. The mass, m , of the object is related to the weight of the object, w as follows:

$$w = mg \quad (1)$$

where g is the acceleration of the gravity. In order to keep complications to a minimum for the sake of analysis purposes, it was assumed that the spring obeys Hooke's law, that is, force is proportional to displacement. Namely,

$$f_s = ky \tag{2}$$

Where y is the displacement and k is the stiffness of the spring element. A realistic analysis of the vertical motion of the mass would take into account not only the elastic and gravitational forces, but also, the effects of the friction affecting the system and all other forces that act externally on the suspended mass. Considering the other forces diminishes the amplitude of the vibration. In order to keep the analysis as simple as possible, let us do not take them into the account now and talk about the details later.

Observing that $m = w/g$ and applying Newton's second law of motion, (*force = mass * acceleration*) to the system gives a very popular equation:

$$\frac{w}{g} \frac{d^2y}{dt^2} + ky = 0 \tag{3}$$

This is an equation of harmonic motion and its solution was discussed in almost every differential equation book (Halliday et al., 2007; Zill & Cullen, 2006). The solution for such a system is:

$$y = A \cos \sqrt{\frac{kg}{w}} t + B \sin \sqrt{\frac{kg}{w}} t \tag{4}$$

If the term $\sqrt{\frac{kg}{w}}$ is set to be ω , then Equation 4 can be written by using a periodic function, as used above, in order to have a more compact form as follows:

$$y = A \cos \omega t + B \sin \omega t \tag{5}$$

Regardless of the values of A and B, that is, regardless of how the system is set in motion, Equation 4 describes the periodic motion with the period of

$$2\pi \sqrt{\frac{w}{kg}} \tag{6}$$

or frequency of

$$\frac{1}{2\pi} \sqrt{\frac{kg}{w}} \tag{7}$$

Whether there is friction in the system or not, the quantity $\frac{1}{2\pi} \sqrt{\frac{kg}{w}}$ is called the *natural frequency*, (ω_n), of the system because this is the frequency at which the spring-mass system would vibrate naturally if no frictional or non-elastic forces other than gravity were present in a given system.

As mentioned earlier, the process by which free vibration diminishes in the amplitude is called the damping effect. If the damping effect presents in the system, the energy of the vibrating system will be dissipated by various mechanisms affecting the system, and often,

more than one mechanism could be present in the system at the same time. In such a system, the damping force, f_D , is related to the velocity across the linear viscous damper by the following equation:

$$f_D = c \frac{dy}{dt} \quad (8)$$

where the constant, c , is the viscous damping coefficient and has the unit of *force*time/length*. It is important to get the correct damping factor for a given system. Unfortunately, unlike the stiffness of the spring, the damping coefficient cannot be calculated from the dimensions of the structure or the size of the structural elements. Therefore, it should be evaluated from the vibration experiments on actual structures in order to get a precise coefficient of damping ratio for minimizing the effects of mechanical vibration to the measurement instruments.

This system is usually called a mass-spring-damp system, and its governing equation can be written according to the Newton's second law of motion as follows:

$$\frac{w}{g} \frac{d^2y}{dt^2} + c \frac{dy}{dt} + ky = p(t) \quad (9)$$

where $p(t) = p_0 \sin(\omega t)$ is the function of the external force (from ground vibration, etc.) acting to the system. The nature of the free motion of the system will depend on the roots of the related characteristic equation of the second order differential equation given in the equation. The characteristic equation for this second order differential equation is given as follows:

$$-\frac{cg}{2w} \mp \frac{g}{2w} \sqrt{c^2 - \frac{4kw}{g}} \quad (10)$$

It is clear that g , k , and w are all positive quantities and c is a non-negative and real number. Therefore, the characteristic of the solution of this second order differential equation depends upon the term $\sqrt{c^2 - \frac{4kw}{g}}$. It is clear that there are three possibilities depending upon the values of k , w , and c , for the solution of this second order differential equation namely,

$$\sqrt{c^2 - \frac{4kw}{g}} \begin{cases} > 0 \\ = 0 \\ < 0 \end{cases} \quad (11)$$

If $\sqrt{c^2 - \frac{4kw}{g}} > 0$, there is a relatively large amount of friction, and, naturally enough, the system or its motion is said to be over-damped. In this case, the roots of the characteristic equation are real and unequal. The general solution is given by:

$$y = Ae^{m_1 t} + Be^{m_2 t} \quad (12)$$

where both of the roots of the second order differential equation, m_1 and m_2 , are negative.

Thus, y approaches zero as time increases indefinitely. If $\sqrt{c^2 - \frac{4kw}{g}} = 0$, it is at the

borderline in which the roots of the characteristic equation are equal and real. In this case, the free motion can be expressed as follows:

$$y = Ae^{mt} + Bte^{mt} = (A + Bt)e^{mt} \tag{13}$$

From the equation $\sqrt{c^2 - \frac{4kw}{g}}$, critical damping can be defined as:

$$c_{cr} = \frac{2w}{g} \omega_n = 2m\omega_n \tag{14}$$

Then, the damping ratio, ζ is defined as:

$$\zeta = \frac{c}{2m\omega_n} = \frac{c}{c_{cr}} \tag{15}$$

If $\sqrt{c^2 - \frac{4kw}{g}} < 0$, the motion is said to be under-damped. The roots of the characteristic equation are then the conjugate complex numbers given by:

$$m_1, m_2 = -\frac{cg}{2w} \mp i \sqrt{\frac{kg}{w} - \frac{c^2g^2}{4w^2}} \tag{16}$$

where $i = \sqrt{-1}$. Then, the general solution for the differential equation is given by:

$$y = Ae^{-\frac{cg}{2w}t} \cos \sqrt{\frac{kg}{w} - \frac{c^2g^2}{4w^2}} t + Be^{-\frac{cg}{2w}t} \sin \sqrt{\frac{kg}{w} - \frac{c^2g^2}{4w^2}} t \tag{17}$$

By defining some of the terms in the equation given above differently and using some trigonometric identities, the solution of the equation may be written in a more compact form as follows:

$$\sqrt{\frac{kg}{w} - \frac{c^2g^2}{4w^2}} = \omega_n \sqrt{1 - \zeta^2} = \omega_D \tag{18}$$

The differential equation is solved, subject to the initial conditions $y = y(0)$, and $\frac{dy}{dt} = \frac{dy}{dt}|_{y=0}$. The particular solution of such a system is given by:

$$y_p = \frac{p_0}{k \left[1 - \frac{\omega^2}{\omega_n^2} \right]^2 + \left[2\zeta \left(\frac{\omega}{\omega_n} \right) \right]^2} \left[\left(1 - \left(\frac{\omega}{\omega_n} \right)^2 \right) \cos(\omega t) - \frac{2\zeta\omega}{\omega_n} \sin(\omega t) \right] \tag{19}$$

By setting $C = \frac{p_0 \left(1 - \frac{\omega^2}{\omega_n^2} \right)}{k \left[1 - \frac{\omega^2}{\omega_n^2} \right]^2 + \left[2\zeta \left(\frac{\omega}{\omega_n} \right) \right]^2}$ and $D = \frac{-2p_0\zeta \frac{\omega}{\omega_n}}{k \left[1 - \frac{\omega^2}{\omega_n^2} \right]^2 + \left[2\zeta \left(\frac{\omega}{\omega_n} \right) \right]^2}$, the Equation 19 can be written as

$$y_p = C \sin(\omega t) + D \cos(\omega t) \tag{20}$$

However, the complete solution of Equation 9 consists of transient and steady parts given as follows:

$$y(t) = e^{-\zeta \omega_n t} \underbrace{(A \cos \omega_D + B \sin \omega_D)}_{\text{transient}} + \underbrace{C \sin \omega t + D \cos \omega t}_{\text{steady state}} \quad (21)$$

where the constant A and B can be determined in terms of the initial displacement and initial velocity. The steady state deformation of the system, due to harmonic force given in Equation 19 can be rewritten as follows:

$$y(t) = y_0 \sin(\omega t - \phi) = \frac{p_0}{k} R_d \sin(\omega t - \phi) \quad (22)$$

where $y_0 = \sqrt{C^2 + D^2}$ and $\phi = \tan^{-1}\left(\frac{D}{C}\right)$. Substituting for C and D gives deformation response factor, R_d , and phase angle, ϕ .

$$R_d = \frac{1}{\sqrt{\left[1 - \frac{\omega^2}{\omega_n^2}\right]^2 + \left[2\zeta\left(\frac{\omega}{\omega_n}\right)\right]^2}} \quad (23)$$

and

$$\phi = \tan^{-1}\left(\frac{2\zeta\frac{\omega}{\omega_n}}{1 - \frac{\omega^2}{\omega_n^2}}\right) \quad (24)$$

Differentiating Equation 22 gives the velocity as follows:

$$\frac{dy(t)}{dt} = \frac{p_0}{\sqrt{km}} R_v \cos(\omega t - \phi) \quad (25)$$

where R_v is the velocity response factor and related to R_d by

$$R_v = \frac{\omega}{\omega_n} R_d \quad (26)$$

After applying the basic definitions and solutions of the differential equation for a mass-spring-damper system, force transmission and vibration isolation can be taken into account as follows: Consider the mass-spring-damper system (The system is the instrument itself and any foundation making the total weight higher), shown in Figure 3 subjected to a harmonic force. The force transmitted to the base is given by:

$$f_T = f_s + f_D = ky(t) + c \frac{dy(t)}{dt} \quad (27)$$

Substituting Equation 22 for $y(t)$ and Equation 25 for $\frac{dy(t)}{dt}$, and using Equation 26 give:

$$f_T(t) = (y_{st})_0 R_d [k \sin(\omega t - \phi) + c\omega \cos(\omega t - \phi)] \quad (28)$$

The maximum value of $f_T(t)$ over t is:

$$(f_T)_0 = (y_{st})_0 R_d \sqrt{k^2 + c^2 \omega^2} \tag{29}$$

which after using $(y_{st})_0 = \frac{p_0}{k}$ and $\zeta = \frac{c}{2m\omega_n}$ can be expressed as:

$$\frac{(f_T)_0}{p_0} = R_d \sqrt{1 + \left(2\zeta \frac{\omega}{\omega_n}\right)^2} \tag{30}$$

Substituting Equation 23 gives an equation for the ratio of the maximum transmitted force to the amplitude p_0 of the applied force, known as the transmissibility (TR) of the system for a mass-spring-damper application:

$$TR = \sqrt{\frac{1 + \left(2\zeta \frac{\omega}{\omega_n}\right)^2}{\left[1 - \frac{\omega^2}{\omega_n^2}\right]^2 + \left[2\zeta \left(\frac{\omega}{\omega_n}\right)\right]^2}} \tag{31}$$

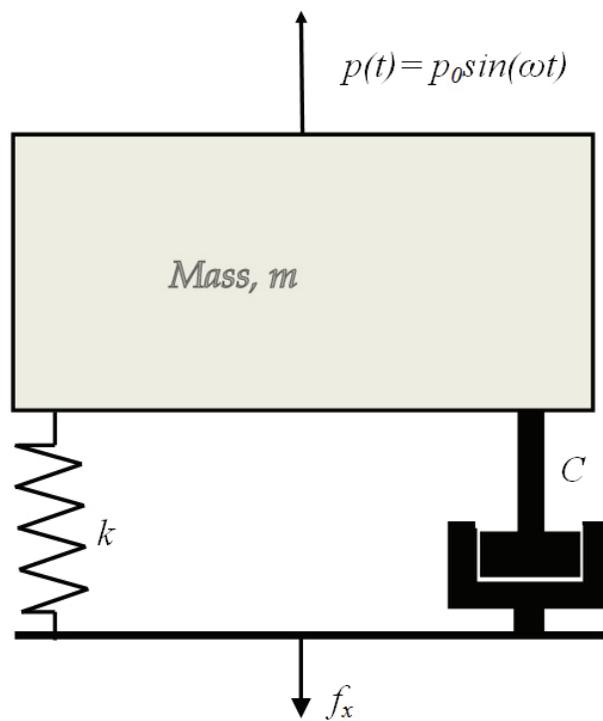


Fig. 3. Simple spring-mass-damper system

The transmissibility is plotted in Figure 4 as a function of the frequency ratio $\frac{\omega}{\omega_n}$ for several values of the damping ratio, ζ . For the transmitted force to be less than the applied force, the stiffness of the support system, and hence, the natural frequency, should be small enough so that the ratio of $\frac{\omega}{\omega_n}$ should be bigger than $\sqrt{2}$ as seen in Figure 4. No damping is desired in the support system because, in this frequency range, damping increases the transmitted force. This implies a trade-off between a soft spring material to reduce the transmitted force and an acceptable static displacement. If the excitation

frequency, ω , is much smaller than the natural frequency, ω_n , of the system, (*i.e.*: the mass is static while the ground beneath it is dynamic). This is the concept underlying isolation of a mass from a moving base by using a very flexible support system. For example, instruments or even buildings have been mounted on natural rubber bearings in order to isolate them from the ground-borne, vertical vibration (typically, with frequencies that range from 25 to 50 Hz) due to the rail traffic (Bozorgnia & Bertero, 2004; Chen & Lui, 2005). It would be also advisable to use rubber like material on the testing instruments in order to diminish the effects of the ground vibrations.

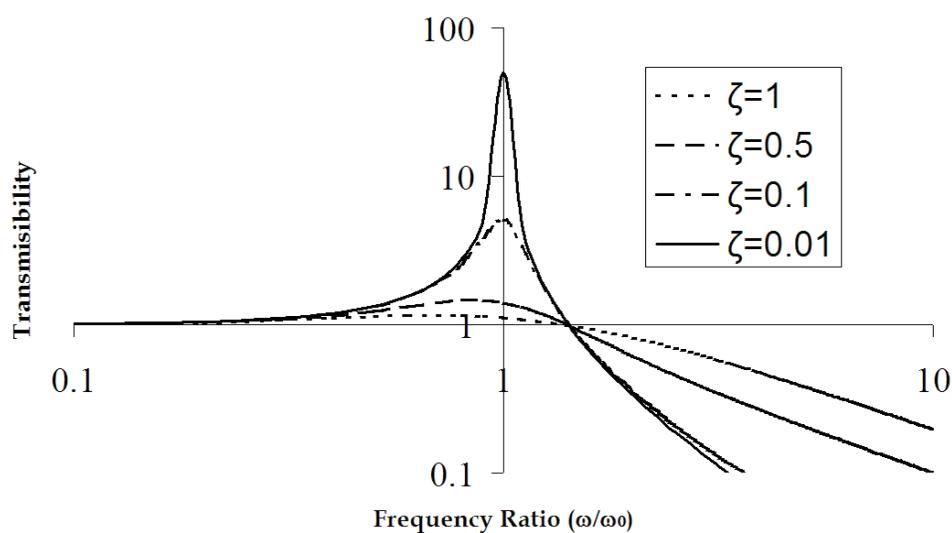


Fig. 4. Transmissibility for harmonic excitation for various damping factors (Axes are in logarithmic scale).

Figure 5 shows a typical building vibration effect on the acquired signal from an instrument. In order to reduce the amount of vibration that is transmitted to the instrument, natural rubber-like materials, such as tennis balls, are the appropriate choice. Figure 6 shows the effect of vibration dampers on the instrument. As seen from the graph, the usage of vibration damper reduces the effects of mechanical vibrations significantly. Some further improvements can be achieved by precisely calculating (and if necessary modifying) the stiffness of the insulation material taking into consideration the low damping coefficient or increasing the weight of instrument.

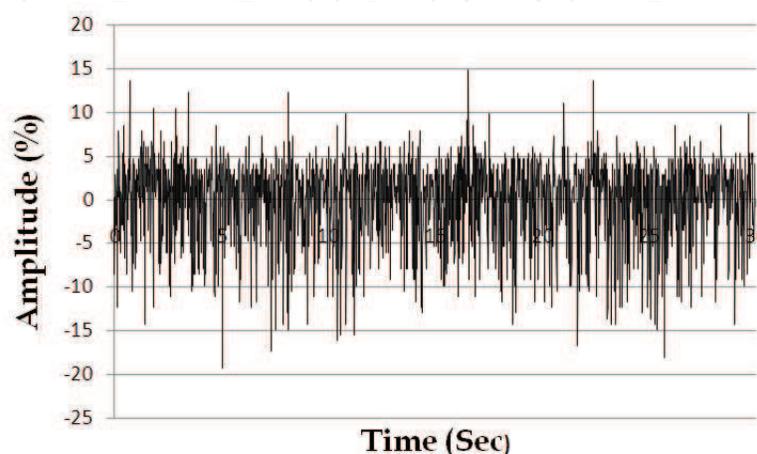


Fig. 5. Effects of building vibration on the acquired signal before using damper

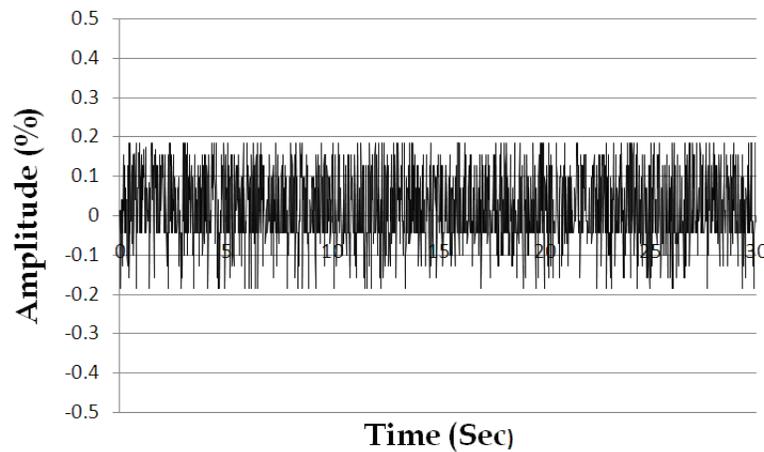


Fig. 6. Effects of building vibration on the acquired signal after using damper

In the analysis of the ground vibration, as seen, only vertical vibration of the building is considered. However, it is clearly known that buildings are exciting in three dimensions. As seen in the analysis, it was assumed that only vertical vibration had a significant effect on the data. This assumption may introduce some experimental errors into the measurement. However, a significant drop in the amplitude of the noise transmitted suggested that either the tennis balls were also eliminating some of the vibration effects coming from the other directions, or the vibration coming from the other directions did not have a significant effect on the signal. Therefore, vibration effects from other directions were not investigated further in this analysis.

In the engineering view of the problem, the ground vibration problem has a crucial effect on measurement instruments. Therefore, it would be advisable to take all of the necessary precautions in order to reduce the amount of transmitted ground vibration to the minimum level as much as possible by using damper systems which make the ratio of \ddagger/\ddagger_n is bigger than $\sqrt{2}$.

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The book reveals many different aspects of motion control and a wide multiplicity of approaches to the problem as well. Despite the number of examples, however, this volume is not meant to be exhaustive: it intends to offer some original insights for all researchers who will hopefully make their experience available for a forthcoming publication on the subject.

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